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PRESSURE SURGE CONTROL AT TRACY PUMPING PLANT

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HYDRAULICS DIVISION

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PRESSURE SURGE CONTROL AT TRACY PUMPING PLANT

John Parmakian[1]

SYNOPSIS

This paper describes the method of waterhammer analysis and control used in the design of the large pump installation at Tracy Pumping Plant, California, for the condition of power failure at the pump motors. It includes a summary of the major features of this installation and a comparison between the observed and computed transient effects. Useful charts for obtaining the approximate waterhammer effects in any pump discharge line without control valves due to a power failure are also presented.

NOMENCLATURE

The following letter symbols are used in this paper:

A	cross sectional area of pipe (ft.^{2})
A_v	open area through butterfly valve (ft.^{2})
a	velocity of waterhammer pressure wave (ft. per sec.)
α	ratio of pump speed at any time to rated pump speed; $\alpha = \frac{N}{N_R}$
β	ratio of pump input torque for any given speed to pump input torque at rated speed and head; $\beta = \frac{M}{M_R}$
C_d	coefficient of discharge of butterfly valve
g	acceleration due to gravity (ft. per sec.^{2})
H_o	pumping head for initial steady pumping conditions (ft.)
H_R	rated pumping head (ft.)
H	pressure head for surge conditions measured above the pump intake water surface elevation (ft.)
H_f	head loss in discharge line due to friction (ft.)
h	ratio of pressure head for surge conditions to rated pumping head; $h = \frac{H}{H_R}$
h_f	ratio of friction head loss to rated pumping head; $h_f = \frac{H_f}{H_R}$
I	mass moment of inertia of rotating parts (lb. ft. sec.^{2})

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K_1 defined by equation 6(sec.$^{-1}$)

L length of pipe (ft.)

$\Sigma\frac{L}{a}$ wave travel time of discharge line (sec.)

M pump input torque corresponding to a given speed and head (lb. ft.)

M_R pump input torque at rated speed and head (lb. ft.)

N pump speed at any time (RPM)

N_R rated pump speed (RPM)

η_R pump efficiency at rated speed and head

Q pump discharge at any time (ft.^{3} per sec.)

Q_o pump discharge for initial pumping conditions (ft.^{3} per sec.)

Q_R pump discharge at rated speed and head (ft.^{3} per sec.)

ρ a pipeline constant as defined by equations (12) or (15)

θ angle between axis of discharge line and the butterfly valve leaf

t time at the instant under consideration during the variable flow conditions: the time t = 0 is taken just prior to power failure (sec.)

τ function of time, equal to the flow through the butterfly valve divided by Q_o

V velocity of water in the discharge line for surge conditions at any time t (ft. per sec.)

V_o velocity of water in the discharge line for initial steady conditions (ft. per sec.)

v ratio of velocity of water for surge conditions to that for initial steady conditions $v = \frac{V}{V_o} = \frac{Q}{Q_o}$

w specific weight of water (lb. per ft.^{3})

ω angular velocity of pump and motor shaft; $\omega = \frac{2\pi N}{60}$ (rad. per sec.)

WR^2 flywheel effect, or moment of inertia in terms of the weight of rotating parts of motor, pump, and entrained water (lb. ft.^{2})

INTRODUCTION

Tracy Pumping Plant is located in the Central Valley Project near Tracy, California. Its function is to pump water from the lower delta region of the Sacramento River to the Delta Mendota Canal. This canal flows along the west side of the San Joaquin Valley for a distance of over 100 miles and supplies water to an extensive irrigation system. The basic data for this plant is shown in Table 1 and its general arrangement is shown in Figures 1 and 2. Briefly, the pumping plant consists of six large pumping units with each pair of pumps connecting through a wye branch to a 15-foot diameter reinforced concrete pipe which extends nearly a mile in length to the upper canal. Each pump is driven by a 22,500-horsepower motor and delivers 767 cubic feet of water per second to the canal under a rated head of 197 feet. Each pump has a butterfly valve on the discharge side of the pump which closes at a variable rate under the action of a servomotor when a power failure occurs at the pump motors.

Transient Conditions Due To Power Failure

There are several operating conditions at Tracy Pumping Plant which are capable of producing substantial pressure changes in the discharge lines. The most serious of these operating conditions is the sudden stoppage of the motors due to a power failure. Figure 3 shows the time history of the pressure, flow, and speed changes at the pumping plant due to a simultaneous power failure at the motors of two pumps on the same discharge line. The transient phenomena is as follows: When the power supply to the pump motors is suddenly cut off, the only energy that is left to drive each pump in the forward direction is the kinetic energy of the rotating elements of the motor and pump and the entrained water in the pump. Since this energy is small when compared to that required to maintain the flow against the discharge head, the reduction in pump speed is very rapid. As the pump speed reduces, the flow of water in the discharge line adjacent to the pump is also reduced. As a result of these rapid flow changes at the pump, waterhammer waves of subnormal pressure are formed in the discharge line at the pump. These subnormal pressure waves move rapidly up the discharge line to the discharge outlet where a wave reflection occurs. Soon the speed of the pump is reduced to a point where no water can be delivered against the existing head and the flow through the pump reverses although the pump may be still rotating in the forward direction. The pump speed now reduces more rapidly and soon the pump reverses in rotation. However, it is prevented from reaching maximum runaway speed in reverse by the closure of the butterfly valve which restricts the reverse flow through the pump. This reduction in the reverse flow produces a pressure rise at the butterfly valve and along the length of the discharge line.

In order to determine the transient hydraulic conditions at the pump and discharge line due to a power failure, four effects must be considered. These are the pump and motor inertia, the pump characteristics, the butterfly valve characteristics, and the waterhammer wave phenomena in the discharge line. The effect of the pump and motor inertia is obtained from the inertia equation. This equation defines the relation between the pump speed and torque at a given instant of time in terms of the kinetic energy of the rotating system. The pump characteristics are obtained from a complete pump characteristics diagram. This diagram defines the manner in which the pump torque and speed vary with the head and discharge throughout the range of operation as a pump, energy dissipator, and turbine. The butterfly valve characteristics are defined by its rate of closure under the action of the servomotor and by the flow through the valve at any time during the closing stroke. Finally, the effects of the waterhammer wave phenomena in the discharge line are obtained from the waterhammer equations. These equations define the relations between the head and flow in the discharge line under the action of the waterhammer waves.

Inertia Equation

When the power to the pump motor is suddenly cut off, the deceleration of the pump at any instant depends upon the flywheel effect of the rotating parts of the pump and motor and the instantaneous torque exerted by the pump impeller. For a rotating system, the accelerating torque is equal to the product of the mass moment of inertia of the rotating system and the angular

acceleration. Following a power failure at the pump motor, the decelerating torque on the rotating system corresponds to the pump input torque prior to power failure. If the decelerating torque is taken as positive,

$$M = -I\frac{d\omega}{dt} = -\frac{WR^2}{g} \cdot \frac{d\omega}{dt} . \qquad (1)$$

Since the pump torque varies with the speed, this equation is applicable for specific instants of time only. For a small time interval $\Delta t = t_2 - t_1$, this equation is written approximately as follows:

$$\frac{M_1+M_2}{2} = -\frac{WR^2}{g}\frac{(\omega_2-\omega_1)}{\Delta t} = \frac{2\pi\, WR^2}{60g} \cdot \frac{(N_1-N_2)}{\Delta t} . \qquad (2)$$

This equation is written with the ratios $\alpha = \frac{N}{N_R}$ and $\beta = \frac{M}{M_R}$ as follows:

$$\alpha_1 - \alpha_2 = \frac{15g\, M_R}{\pi WR^2 N_R}(\beta_1 + \beta_2)\ \Delta t. \qquad (3)$$

The decelerating torque at the rated head and speed of the pump is:

$$M_R = \frac{60w\, H_R\, Q_R}{2\pi\, N_R\, \eta_R} . \qquad (4)$$

Then

$$\alpha_1 - \alpha_2 = K_1(\beta_1 + \beta_2)\ \Delta t, \qquad (5)$$

where

$$K_1 = \frac{450\, g\, w\, H_R\, Q_R}{\pi^2\, WR^2\, \eta_R\, N_R^2} = \frac{91{,}600\, H_R\, Q_R}{WR^2\, \eta_R\, N_R^2} . \qquad (6)$$

Pump Characteristics

The pump performance data supplied by the pump manufacturer for the Tracy pumps are shown in Figure 4 and include the head, brake-horsepower, and efficiency plotted against the pump discharge. These data are converted to a family of torque and speed curves on an h, v diagram in the following manner: Referring to Figure 5, the point corresponding to the rated head and discharge of the pump has the coordinates h = 1, v = 1. The curves for $\alpha = 1$ and $\beta = 1$ also pass through this point on the diagram, since these ratios are defined in terms of rated values. Other values of β on the curve for $\alpha = 1$ are determined by computing values of h and v from the brake-horsepower curve given in Figure 4. Points on the diagram for other values of α and β are then determined from the laws of homologous pump operation. For example, for a given pump, H and M vary with N^2 while N and $H^{\frac{1}{2}}$ vary with Q. To use these relations on the h, v diagram a parabola of the type $h = K_2 v^2$ is passed through a known point on the diagram such as that shown in Figure 6 at $\alpha = 1.0$, $\beta = 0.8$. The vertical distance h = 1.12 at this point is then divided into eight equal parts, and the points corresponding to $\beta = 0.7, 0.6$, etc. are located on the parabola at ordinate values of h = 7/8(1.12), 6/8(1.12), etc. Similarly, points corresponding to $\alpha = 0.9, 0.8$ are located on the same parabola at values of $h = 1.12(0.9)^2$, $1.12(0.8)^2$, etc. A smooth curve is then drawn through the points on the various parabolas corresponding to each value of α and β. Figure 5 shows families of torque and speed curves which were constructed in this manner.

The pump characteristics shown in Figure 5 define the manner in which the pump torque and speed vary with the head and discharge for the zone of normal pump operation. For the condition of power failure at the pump motors these characteristics are sufficient for determining the minimum transient pressures and likelihood of water column separation at all points in the discharge line. However, after the flow reverses through the pump and prior to the control of the reverse flow by the butterfly valve, additional pump characteristics are required for the zone of energy dissipation in which the pump is running in the forward direction with the flow through the pump in reverse, and for the zone of turbine operation in which the flow through the pump and the pump rotation are both in reverse. The characteristics of the Tracy pumps for these two zones of abnormal operation were not available but were estimated from the complete pump characteristics shown in Figures 7a, b, and c for another double volute pump.

Butterfly Valve Characteristics

The butterfly valves used at Tracy Pumping Plant have a wide open area of about 47.9 square feet and an 82 degree angle of rotation of the valve leaf. Upon power failure the first 55 degree angle of closure of the valve leaf is accomplished in 14 seconds under the action of the servomotor, and the remaining 27 degree angle of closure is then completed at a slower rate in 46 additional seconds. The open area of the butterfly valve normal to the axis of the pipe is very nearly

$$A_v = 47.9 \left(1 - \frac{\sin \theta}{\sin 82^\circ}\right), \qquad (7)$$

and the flow through the butterfly valve is

$$Q = C_d A_v \sqrt{2gH}. \tag{8}$$

When the variable rate of closure is considered and a coefficient of discharge of 0.7 is used, the flow ratio is defined by the equation,

$$v = \tau \sqrt{h} \tag{9}$$

where τ, a function of time, is shown in Figure 8.

Waterhammer Equations

The waterhammer equations for the pump discharge line shown in Figure 9 are as follows:

$$h_{B_1 t_1} - h_{C_1 t_2} = - 2\rho_1 \, (V_{B_1 t_1} - V_{C_1 t_2}), \tag{10}$$

$$h_{C_2 t_3} - h_{B_2 t_4} = + 2\rho_2 \, (V_{C_2 t_3} - V_{B_2 t_4}). \tag{11}$$

In these equations $(t_2 - t_1)$ is the wave travel time between B_1 and C_1 for an F type pressure wave and $(t_4 - t_3)$ is the wave travel time between C_2 and B_2 for an f type pressure wave. For a uniform discharge line,

$$2\rho = \frac{aV_o}{gH_o}, \qquad \text{the pipeline constant.} \tag{12}$$

Since the Tracy Pumping Plant discharge line consists of a number of lengths of uniform pipe of characteristics L_1, A_1, a_1,etc. in series, the discharge line is considered to be the same hydraulically as an equivalent uniform pipe line of characteristics L, A, and a, by making the following substitution:

$$\frac{L}{A} = \Sigma\left(\frac{L_n}{A_n}\right) = \frac{L_1}{A_1} + \frac{L_2}{A_2} + \frac{L_3}{A_3} + \ldots\ldots + \frac{L_n}{A_n}, \tag{13}$$

$$\frac{L}{a} = \Sigma\left(\frac{L_n}{a_n}\right) = \frac{L_1}{a_1} + \frac{L_2}{a_2} + \frac{L_3}{a_3} + \ldots\ldots + \frac{L_n}{a_n}, \tag{14}$$

and

$$2\rho = \frac{Q_o \Sigma(\frac{L_n}{A_n})}{gH_o \Sigma(\frac{L_n}{a_n})} \quad . \tag{15}$$

Equations (10) and (11) define the relations between the head and velocity ratios during the transient waterhammer phenomena. On a coordinate system of h and v these equations are represented by straight lines which have a slope of $\pm 2\rho$ as shown in Figure 10. Each waterhammer equation gives one relation between the value of h and v at a point in a uniform section of the discharge line in terms of h and v which are known at another point in the line. In order to determine h and v explicitly, a second independent relation is required. When the pump is controlling the flow, the second independent relation is obtained from the pump-inertia equation with the use of the pump characteristics diagram. When the butterfly valve has closed sufficiently so that it is controlling the reverse flow through the pump, the second independent relation is obtained from the head flow ratios at the butterfly valve as defined by equation (9). These flow relations at the butterfly valve are represented by parabolas on the h, v diagram.

Effect of Pipeline Friction

When two pumps are delivering water to the same discharge line the pipeline friction loss is a maximum and is estimated at about 5.6 feet for pump operation at the rated head. Since this head loss is a small percentage of the pumping head, its effect is approximated in the waterhammer solution by locating a hypothetical obstruction at the pump discharge outlet. This hypothetical obstruction is assumed to have the same head loss as the entire pipeline friction loss. This is represented on a coordinate system of h and v by a head loss curve as shown in Figure 11 where, for forward flow in the zone of normal pump operation the head ratio at the pump discharge outlet at F is,

$$h_F = 1 - \frac{H_f}{H_R}(1 - v^2) = 1 - 0.028\ (1 - v^2) \tag{16A}$$

and for reverse flow through the pump,

$$h_F = 1 - (0.028)\ (1 + v^2) \quad . \tag{16B}$$

Graphical Waterhammer Analysis

For a power failure at the motors of two pumps which are supplying water to the same discharge line at Tracy Pumping Plant,

$$K_1 = \frac{91{,}600 \times 1534 \times 197}{2 \times 4.040 \times 10^6 \times (180)^2 \times 0.88} = 0.1204,$$

as defined by equation (6). The wave travel time for the discharge line

$$\Sigma \frac{L}{a} = 1.75 \text{ seconds and } \Sigma \frac{L}{A} = \frac{5130}{176.7} = 29.0 .$$

Then from equation (15),

$$2\rho = \frac{1534 \times 29.0}{32.2 \times 197 \times 1.75} = 4.0 .$$

In order to determine the pressure changes at the pump and four intermediate points along the discharge line, a time interval $\Delta t = \frac{2L}{5a} = 0.7$ seconds is used in the computations. Then $K_1 \cdot \Delta t = 0.0843$ and $\alpha_1 - \alpha_2$ as defined by equation (5) is

$$\alpha_1 - \alpha_2 = 0.0843 \, (\beta_1 + \beta_2). \qquad (17)$$

The simultaneous solution of the waterhammer equations with the pump inertia equations on the pump characteristics diagram by graphical methods is shown in Figure 11. This solution is completed up to the point at which the butterfly valve closure starts to restrict the flow through the pump. The starting point in Figure 11 is at A_0 which is located at the coordinates h = 1, v = 1. $A_{0.7}$ is then located along the line of slope $2\rho = +4.0$ which passes through A_0 in the following manner:

The location of $A_{0.7}$ is first estimated. Values of α_2 and β_2 at the estimated position of $A_{0.7}$ are then read from the pump characteristics curves. This value of β_2 is used in equation (17) to compute α_2. If the computed value of α_2 does not agree with the value of α_2 read from the pump characteristics curves, the point for $A_{0.7}$ is shifted up or down on the sloping line $2\rho = 4.0$ until these values of α_2 agree. This particular point is found to be located at a value of $\beta = 0.767$ and $\alpha = 0.851$. Other points for $A_{1.4}$, $A_{2.1}$, etc. are determined in a similar manner. The values for several of these points are shown in Table 2 and the graphical waterhammer solution is completed as shown in Figure 11 up to t = 14.0 seconds at which point the butterfly closure starts to restrict the flow.

Parabolas corresponding to the flow through the butterfly valve as defined by equation (9) are shown in Figure 12 for each 3.5 second time interval. The

intersection of the lines of $\pm 2\rho$ slope with the pertinent parabolas and the headloss curve as shown in Figure 12 completes the waterhammer solution due to the butterfly valve closure.

A time history of the head, flow, and speed changes as obtained from the graphical waterhammer solution is shown in Figure 3. In addition to the time history the following limiting values are also obtained from the graphical solution:

(A) Maximum drop in head at the pump = 0.93 H_R

(B) Maximum drop in head at B = 0.80 H_R

(C) Maximum drop in head at C = 0.67 H_R

(D) Maximum drop in head at D = 0.51 H_R

(E) Maximum drop in head at E = 0.30 H_R

(F) Maximum head rise at the butterfly valve = 1.17 H_R

Water Column Separation

The maximum positive and negative pressures obtained from the graphical waterhammer solution are plotted on the discharge line profile in Figure 13 to show the limiting pressures which are exerted on the discharge line due to a power failure at the two pump motors. If the minimum pressure at any point in the pipeline had reached the vapor pressure of water or about 33.4 feet below atmospheric pressure, the waterhammer solution would no longer have been valid and water column separation in the discharge line would have been a possibility. Water column separation sometimes occurs during the initial negative surge waves on long pump discharge lines at high points which are near the hydraulic gradient. Whenever possible, this condition is avoided because of the heavy surge pressures that are sometimes created when the separated water columns rejoin. At Tracy Pumping Plant the computed minimum pressure at the centerline of the discharge line is about 17 feet below atmospheric pressure and hence water column separation is improbable.

Observed and Computed Results

Pressure surge tests were conducted at Tracy Pumping Plant for the condition of power failure at the pump motors to check the adequacy of the pumps, discharge lines, and pressure control equipment, and also to check the accuracy of the waterhammer analyses. The tests included the following conditions of operation:

(A) Two pumps supplying water to the same discharge line when a power failure occurs at both pump motors,

(B) Two pumps supplying water to the same discharge line when a power failure occurs at only one pump motor,

(C) One pump supplying water to the discharge line when a power failure occurs at the pump motor.

In general, agreement between the computed and observed results was very satisfactory. Moreover, there was no evidence of water column separation in the discharge line. A comparison between the observed and computed results for a power failure at two pump motors is shown in Figure 3.

Waterhammer Charts

In obtaining the graphical waterhammer solution for a pump installation subsequent to a power failure at the pump motors, two independent parameters are used with the complete pump characteristics. These are 2ρ, the pipeline constant, and $K_1\frac{2L}{a}$, a constant which includes the effect of the pump and motor inertia and the waterhammer wave travel time of the discharge line. For a given set of pump characteristics, the results of a large number of waterhammer solutions can be shown on charts as indicated in Figures 14a to d. These charts furnish a convenient method for estimating the transient effects due to a power failure at the pump motors when no control valves are present at the pump. Although the charts were obtained by using the complete characteristics of a particular pump operating at the rated head prior to power failure, they are useful for obtaining the approximate waterhammer effects in any pump discharge line.

Use of Surge Tanks and Air Chambers at Tracy Pumping Plant

During the initial design studies of Tracy Pumping Plant surge tanks or air chambers were considered necessary for preventing water column separation and for reducing the waterhammer effects in the discharge line due to a power failure at the pump motors. In the final designs it was ascertained that since water column separation was improbable, surge tanks or air chambers would not be required.

CONCLUSIONS

Pressure surges in pump discharge lines subsequent to a power failure at the pump motors can be determined with satisfactory accuracy if the complete characteristics of the pump and control valves are known. For Tracy Pumping Plant only the pump characteristics for the zone of normal pump operation were obtainable from the pump performance data supplied by the pump manufacturer. This permitted an accurate determination of the waterhammer effects in the discharge line up to the point at which the flow reversed through the pump. However, this was sufficient for determining the possibility of water column separation in the discharge line. Pump characteristics for abnormal operation and the characteristics of the butterfly valve were then estimated for the remainder of the waterhammer solution. This type of waterhammer analysis afforded a large monetary savings for Tracy Pumping Plant by the elimination of expensive surge tanks or air chambers which were initially considered necessary for pressure control.

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Table 1

BASIC DATA FOR TRACY PUMPING PLANT

Type of pump, double volute

Number of pumps in pumping plant, 6

Number of pumps on each discharge line, 2

Number of pump discharge lines, 3

Warranted pump discharge at rated head, 767 cubic feet per second

Rated pumping head, 197 feet

Range of pumping head, 174 to 204 feet

Length of pump discharge lines, 5130 feet

Waterhammer wave travel time in each pump discharge line, $\Sigma \frac{L}{a} = 1.75$ seconds

Diameter of discharge line, 15 feet

Pipe line constant as defined by equation 15, $\rho = 2.0$

Type of discharge conduit, buried reinforced concrete pipe

Motor horsepower rating, 22,500 horsepower

Motor speed, 180 rpm

Motor WR^2, 3,500,000 pound feet^2

WR^2 of pump and entrained water, 540,000 pound feet^2

Pump and motor inertia constant as defined by equation (6), $K_1 = 0.120$

Type of control valve at each pump, butterfly valve

Open area of butterfly valve, 47.9 square feet

Closing rate of butterfly valve leaf, first 55° in 14 seconds, remaining 27° in 46 additional seconds

Type of discharge outlet, siphon, interrupted by power failure

Date pump units placed in operation, 1951

Table 2

PUMP SPEED AND TORQUE RATIOS

Location	α (Speed Ratio)	β (Torque Ratio)	$a_1 - a_2$ (From Curves)	$a_1 - a_2$ (From Equation 17)
A_0	1.000	1.000	0	0
$A_{0.7}$	0.851	0.767	0.149	0.149
$A_{1.4}$	0.736	0.600	0.115	0.115
$A_{2.1}$	0.646	0.467	0.090	0.090
$A_{2.8}$	0.574	0.385	0.072	0.072
$A_{3.5}$	0.515	0.310	0.059	0.059
$A_{4.2}$	0.468	0.245	0.047	0.047
$A_{4.9}$	0.431	0.198	0.037	0.037
$A_{5.6}$	0.400	0.165	0.031	0.031
$A_{6.3}$	0.374	0.140	0.026	0.026

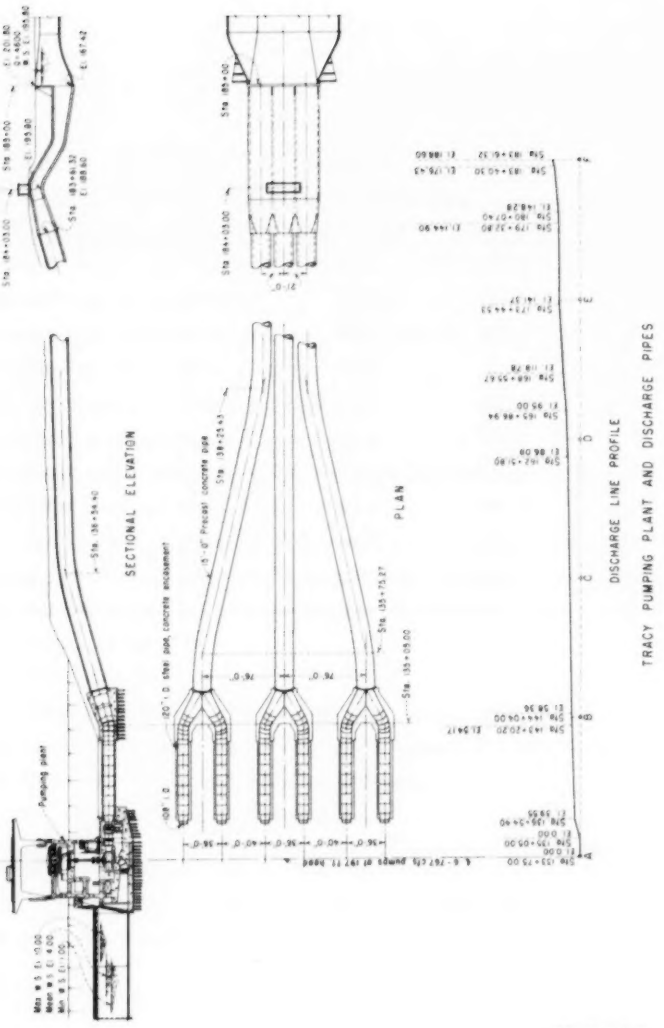

SECTIONAL ELEVATION
PLAN
DISCHARGE LINE PROFILE
TRACY PUMPING PLANT AND DISCHARGE PIPES

FIGURE 1

FIGURE 2

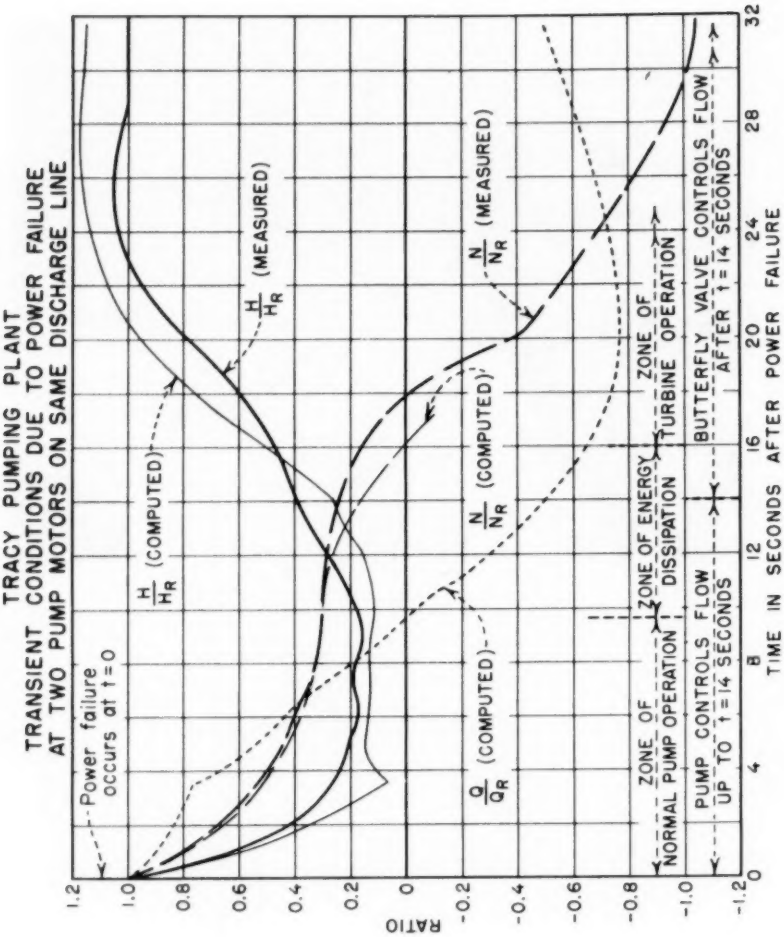

TRACY PUMPING PLANT
TRANSIENT CONDITIONS DUE TO POWER FAILURE
AT TWO PUMP MOTORS ON SAME DISCHARGE LINE
Power failure occurs at t = 0
H/H_R (COMPUTED)
H/H_R (MEASURED)
Q/Q_R (COMPUTED)
N/N_R (COMPUTED)
N/N_R (MEASURED)
ZONE OF NORMAL PUMP OPERATION
ZONE OF ENERGY DISSIPATION
ZONE OF TURBINE OPERATION
PUMP CONTROLS FLOW UP TO t = 14 SECONDS
BUTTERFLY VALVE CONTROLS FLOW AFTER t = 14 SECONDS
RATIO
TIME IN SECONDS AFTER POWER FAILURE
1.2
1.0
0.8
0.6
0.4
0.2
0
-0.2
-0.4
-0.6
-0.8
-1.0
-1.2
0
4
8
12
16
20
24
28
32

FIGURE 3

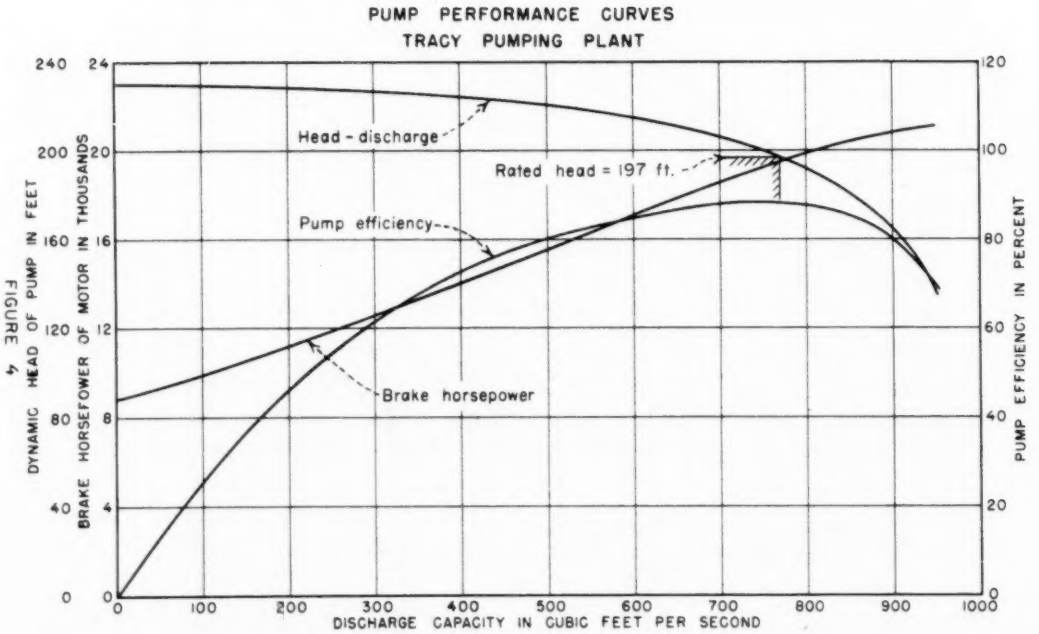

PUMP PERFORMANCE CURVES
TRACY PUMPING PLANT
Head - discharge
Rated head = 197 ft.
Pump efficiency
Brake horsepower
DYNAMIC HEAD OF PUMP IN FEET
BRAKE HORSEPOWER OF MOTOR IN THOUSANDS
PUMP EFFICIENCY IN PERCENT
DISCHARGE CAPACITY IN CUBIC FEET PER SECOND
0
40
80
120
160
200
240
4
8
12
16
20
24
0
20
40
60
80
100
120
0
100
200
300
400
500
600
700
800
900
1000

FIGURE 4

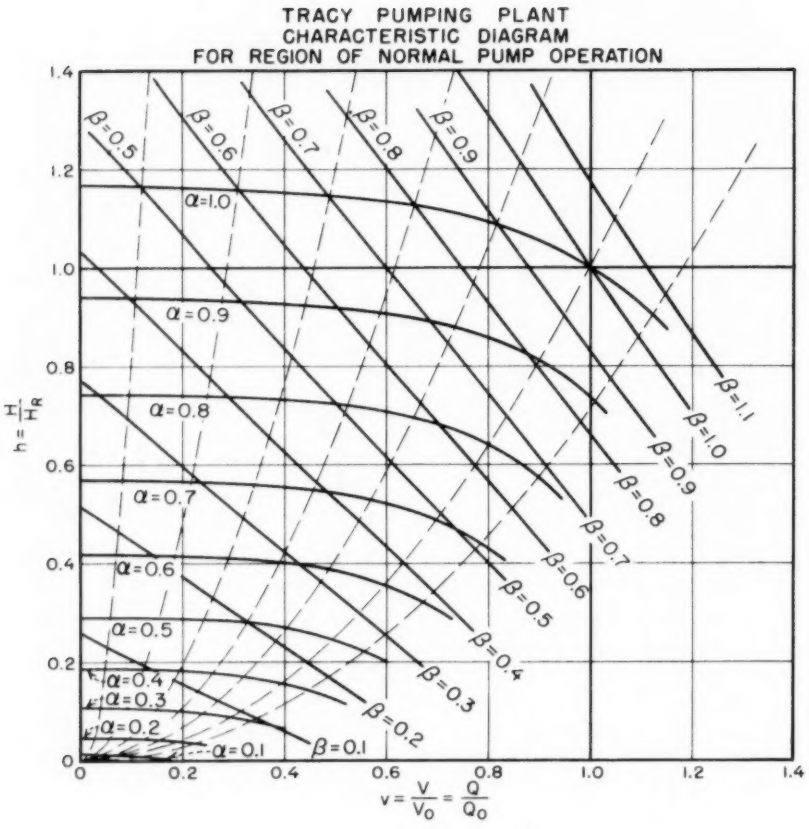

TRACY PUMPING PLANT
CHARACTERISTIC DIAGRAM
FOR REGION OF NORMAL PUMP OPERATION
h = H/H_R
v = V/V_0 = Q/Q_0
β=0.5
β=0.6
β=0.7
β=0.8
β=0.9
β=1.0
β=1.1
α=1.0
α=0.9
α=0.8
α=0.7
α=0.6
α=0.5
α=0.4
α=0.3
α=0.2
α=0.1
β=0.1
β=0.2
β=0.3
β=0.4
β=0.5
β=0.6
β=0.7
β=0.8
β=0.9
β=1.0
β=1.1
0
0.2
0.4
0.6
0.8
1.0
1.2
1.4

FIGURE 5

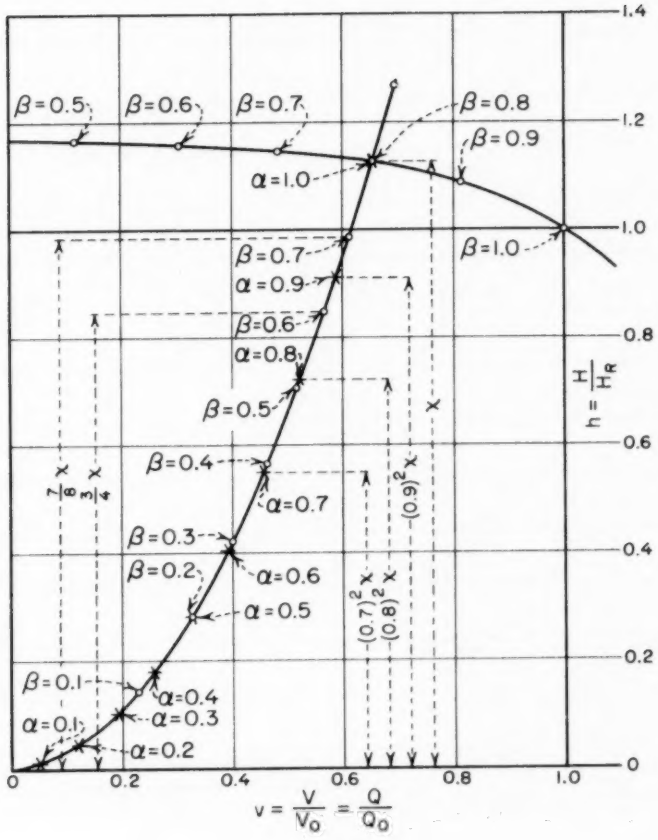

β=0.5
β=0.6
β=0.7
β=0.8
β=0.9
α=1.0
β=1.0
β=0.7
α=0.9
β=0.6
α=0.8
β=0.5
β=0.4
α=0.7
β=0.3
β=0.2
α=0.6
α=0.5
β=0.1
α=0.4
α=0.3
α=0.1
α=0.2
7/8 χ
3/4 χ
χ
(0.9)² χ
(0.8)² χ
(0.7)² χ
h = H/H^α
v = V/V₀ = Q/Q₀
0
0.2
0.4
0.6
0.8
1.0
1.2
1.4

FIGURE 6

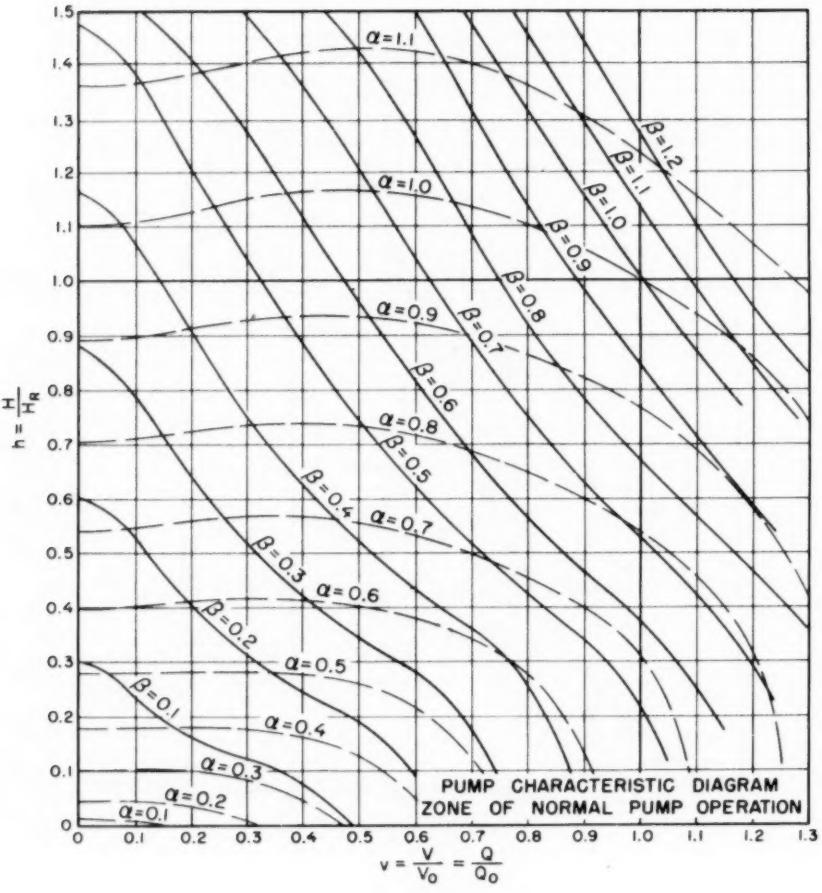

PUMP CHARACTERISTIC DIAGRAM
ZONE OF NORMAL PUMP OPERATION
α=1.1
α=1.0
α=0.9
α=0.8
α=0.7
α=0.6
α=0.5
α=0.4
α=0.3
α=0.2
α=0.1
β=1.2
β=1.1
β=1.0
β=0.9
β=0.8
β=0.7
β=0.6
β=0.5
β=0.4
β=0.3
β=0.2
β=0.1
h = H/H_R
v = V/V_0 = Q/Q_0

FIGURE 7a

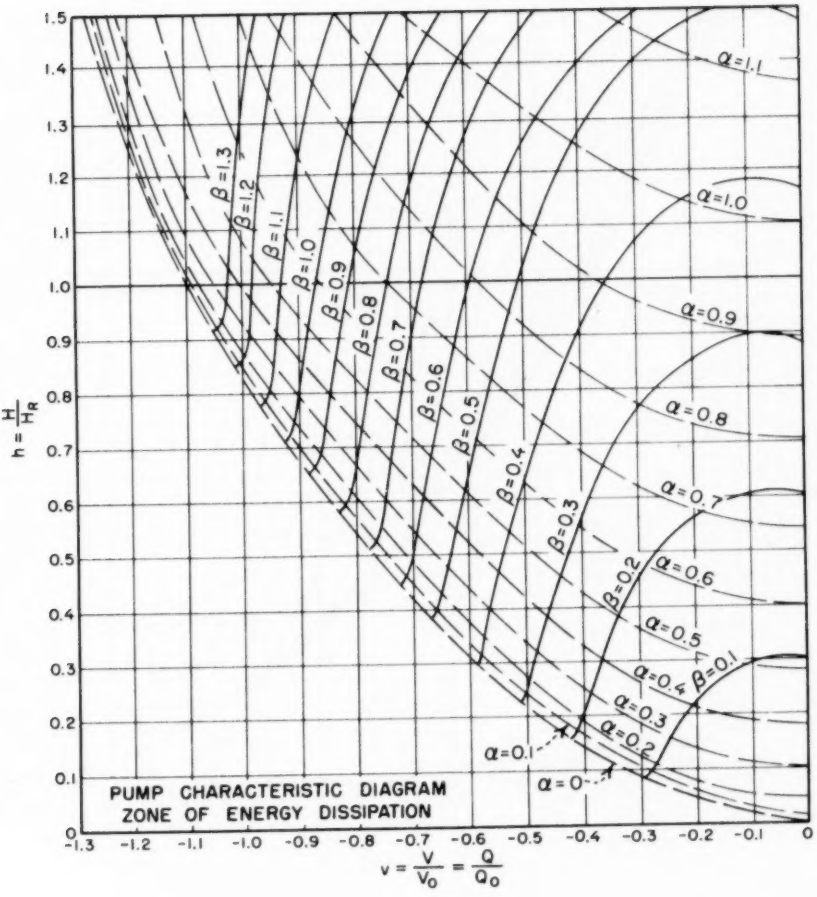

PUMP CHARACTERISTIC DIAGRAM
ZONE OF ENERGY DISSIPATION
h = H/H_R
v = V/V_0 = Q/Q_0
α=1.1
α=1.0
α=0.9
α=0.8
α=0.7
α=0.6
α=0.5
α=0.4
α=0.3
α=0.2
α=0.1
α=0
β=1.3
β=1.2
β=1.1
β=1.0
β=0.9
β=0.8
β=0.7
β=0.6
β=0.5
β=0.4
β=0.3
β=0.2
β=0.1

FIGURE 7b

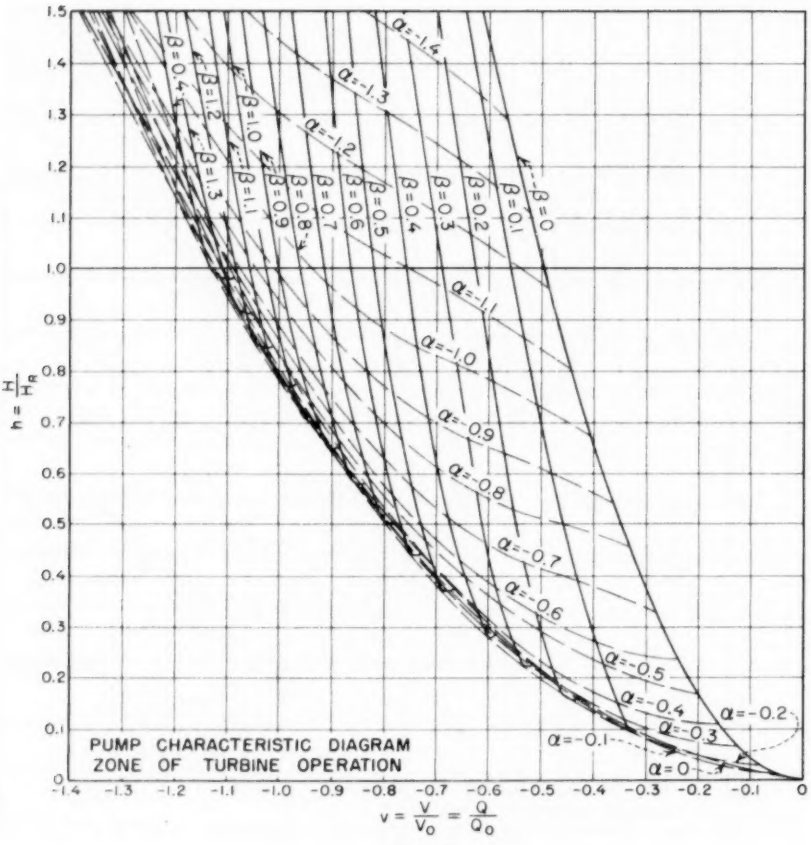
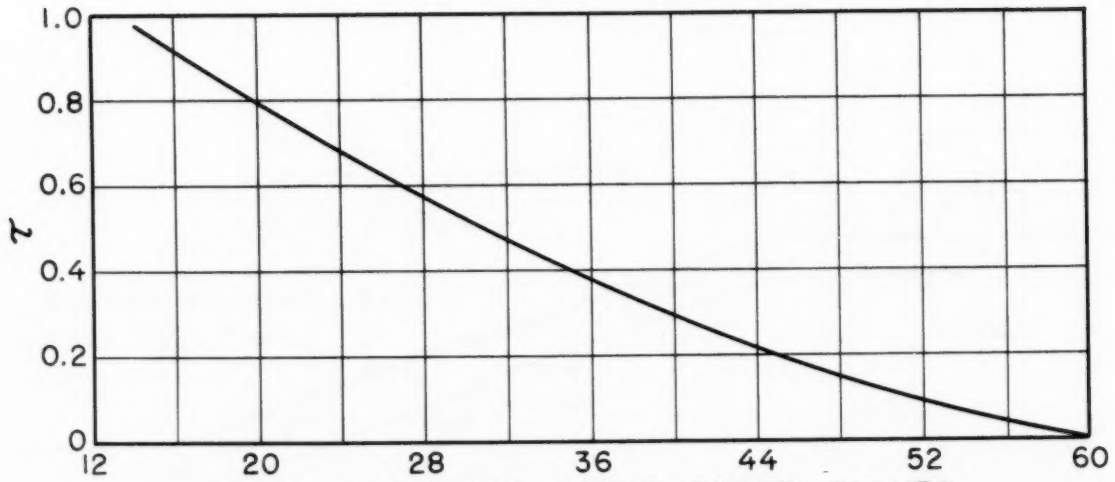

PUMP CHARACTERISTIC DIAGRAM
ZONE OF TURBINE OPERATION
h = H/H_R
v = V/V_0 = Q/Q_0
α=-1.4
α=-1.3
α=-1.2
α=-1.1
α=-1.0
α=-0.9
α=-0.8
α=-0.7
α=-0.6
α=-0.5
α=-0.4
α=-0.3
α=-0.2
α=-0.1
α=0
β=0
β=0.1
β=0.2
β=0.3
β=0.4
β=0.5
β=0.6
β=0.7
β=0.8
β=0.9
β=1.0
β=1.1
β=1.2
β=1.3
β=0.4

FIGURE 7c



BUTTERFLY VALVE CLOSURE RELATION

FIGURE 8

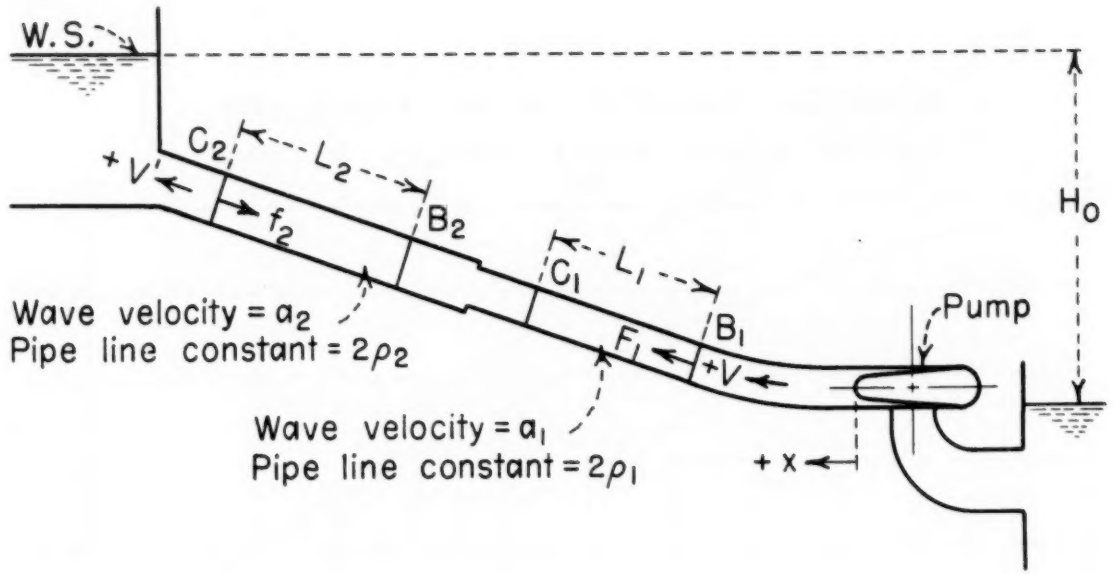

W.S.
C_2
L_2
B_2
C_1
L_1
B_1
$+V$
f_2
F_1
$+V$
H_o
Pump
Wave velocity = a_2
Pipe line constant = $2\rho_2$
Wave velocity = a_1
Pipe line constant = $2\rho_1$
$+x$

FIGURE 9

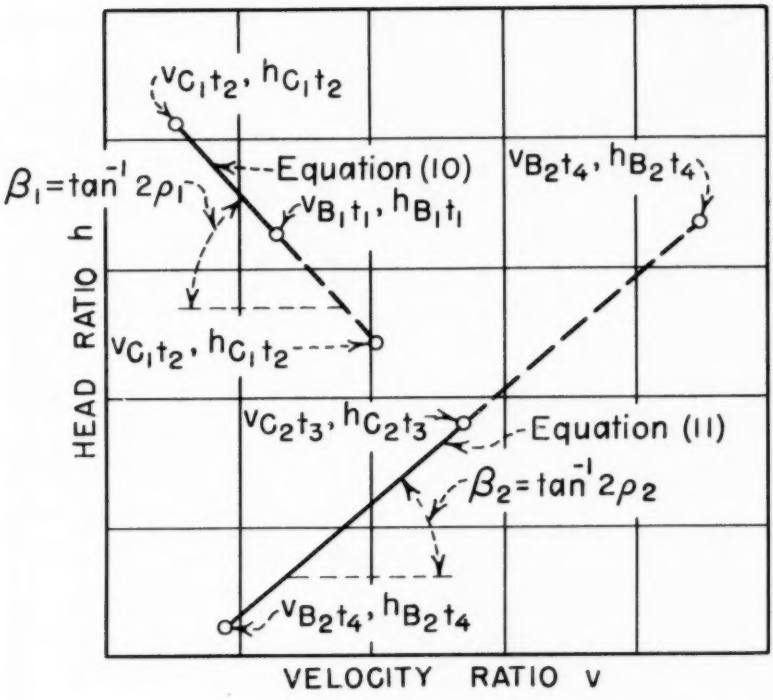

$v_{C_1t_2}, h_{C_1t_2}$
$\beta_1 = \tan^{-1} 2\rho_1$
Equation (10)
$v_{B_1t_1}, h_{B_1t_1}$
$v_{B_2t_4}, h_{B_2t_4}$
$v_{C_1t_2}, h_{C_1t_2}$
$v_{C_2t_3}, h_{C_2t_3}$
Equation (11)
$\beta_2 = \tan^{-1} 2\rho_2$
$v_{B_2t_4}, h_{B_2t_4}$
HEAD RATIO h
VELOCITY RATIO v

FIGURE 10

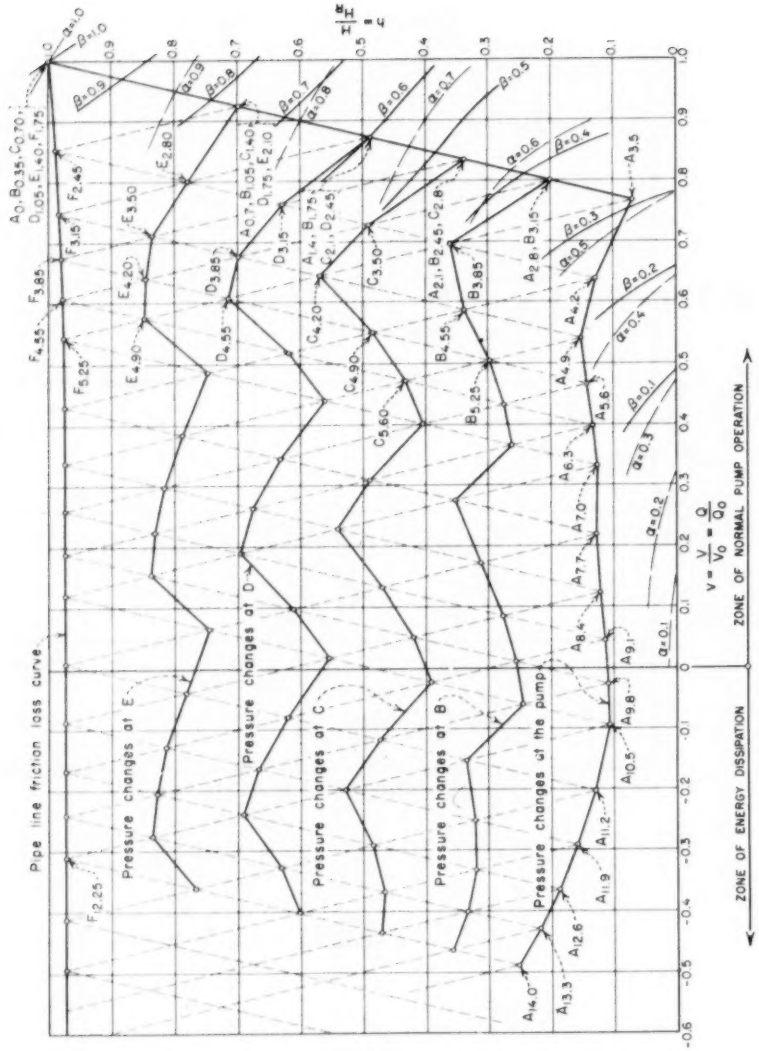

Pipe line friction loss curve
Pressure changes at E
Pressure changes at D
Pressure changes at C
Pressure changes at B
Pressure changes at the pump
ZONE OF ENERGY DISSIPATION
ZONE OF NORMAL PUMP OPERATION
$\frac{H}{H} = 1$
$v = \frac{V}{V_0} = \frac{Q}{Q_0}$

FIGURE 11

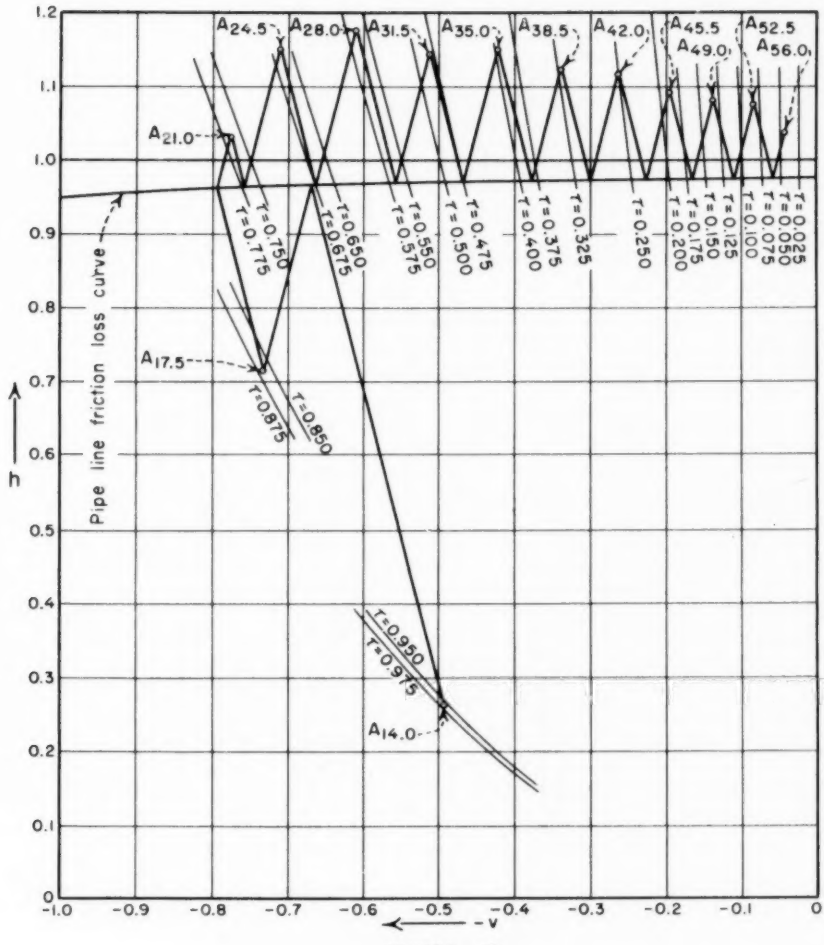

A24.5
A28.0
A31.5
A35.0
A38.5
A42.0
A45.5
A49.0
A52.5
A56.0
A21.0
A17.5
A14.0
τ=0.750
τ=0.775
τ=0.650
τ=0.675
τ=0.550
τ=0.575
τ=0.475
τ=0.500
τ=0.375
τ=0.400
τ=0.325
τ=0.250
τ=0.200
τ=0.175
τ=0.150
τ=0.125
τ=0.100
τ=0.075
τ=0.050
τ=0.025
τ=0.875
τ=0.850
τ=0.950
τ=0.975
Pipe line friction loss curve
h
-v
1.2
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
-1.0
-0.9
-0.8
-0.7
-0.6
-0.5
-0.4
-0.3
-0.2
-0.1
0

FIGURE 12

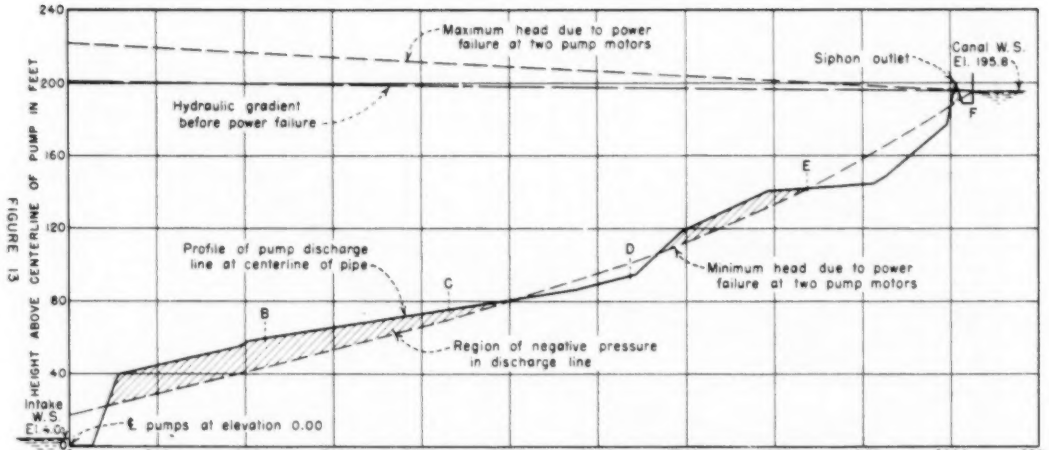
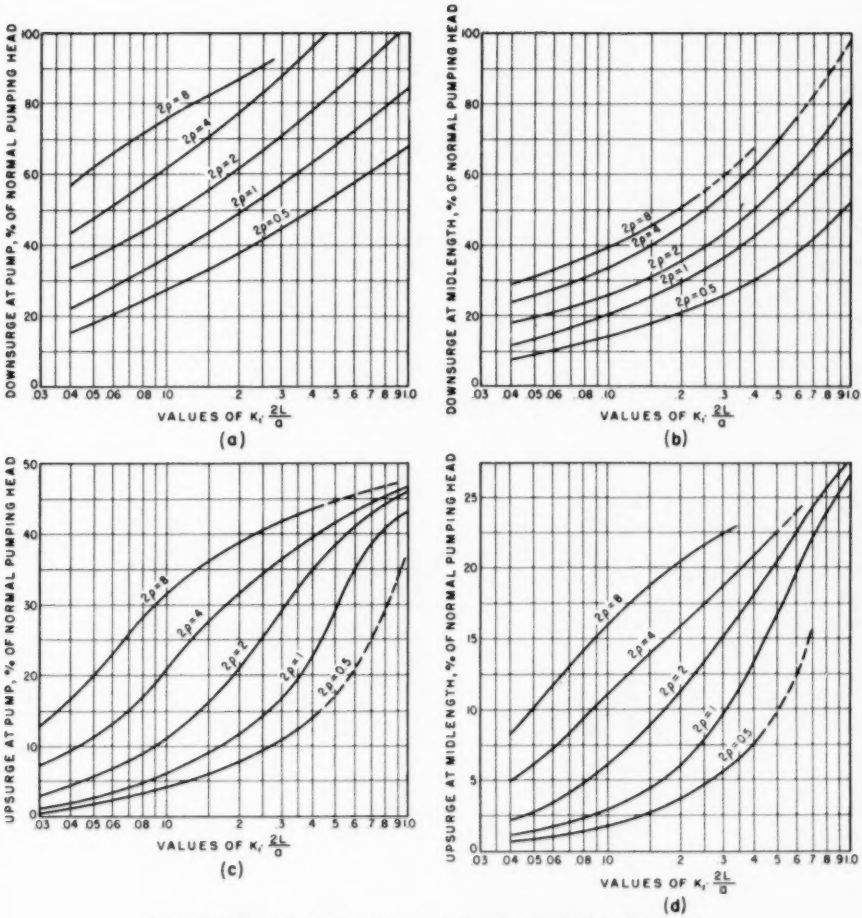


FIGURE 13

TRANSIENT PRESSURE GRADIENTS IN DISCHARGE LINES DUE TO POWER FAILURE

TRACY PUMPING PLANT



WATERHAMMER EFFECTS IN PUMP DISCHARGE LINES DUE TO A POWER FAILURE AT THE PUMP MOTORS

FIGURE 14

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